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MAIN TRENDS IN THE IMPROVEMENT OF MARINE STEAM AUXILIARY TURBIN--ETC(U)
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(Osnovnyye naprovleniya sovershenstvovaniya sudovykh parovykh vspomogatel'nykh turbin)

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MAIN TRENDS IN THE IMPROVEMENT OF MARINE STEAM AUXILIARY TURBINES

[Kurzon, A. G. (Doctor of Technical Sciences, Vlasov, Ye. N. (Engineer); Osnovnyye napravleniya sovershenstvovaniya sudovykh parovykh vspomogatel'nykh turbin; Sudostroyeniye, no. 2, 1964, pp. 21-25; Russian]

For the heat efficiency of a steam turbine plant, the efficiency of the auxiliary machinery drive whose value is chiefly determined by the actual efficiency of the auxiliary turbines is of great importance /1, 2/. At present, the actual efficiency of the better auxiliary turbines has been brought up to 40-45%.

Significant investigations of the supersonic turbine stages have been carried out in the laboratories of several organizations and plants. These investigations mostly involve stages of considerable power with a relatively low degree of expansion of the working body and an intake factor of $\varepsilon = 0.4-1.0$. Of late, investigators have been turning to low-power stages with low admissions. Unfortunately, the theoretical advances have been very slow in penetrating ship planning practice and consequently the available possibilities of increasing the efficiency of the auxiliary turbines are not being used. The purpose of this article is to call attention to these possibilities.

Let us introduce the following symbols: N_e is the effective power, kWt; n is speed, rpm; P_0 is steam pressure before the shut-off valve, kg/cm^2; t_1 is the steam temperature before the nozzles, C°; P_1, P_2 are the steam pressures before the nozzles and after the turbine, kg/cm^2; H_{ad} is the adiabatic drop, kcal/kg; u is the peripheral speed of average diameter disc, m/sec; c_0 is the theoretical steam outlet velocity from nozzles, m/sec; c_2 is the steam velocity at outlet from stage, m/sec; ε is the intake factor (admission); D is the average blading diameter, mm; G is the designed steam rate, kg/hr; M_{c1} is the Mach number at the outlet from the nozzle system; M_{w1} is the M number with respect to the relative velocity of steam to the blades; a_1 angle steam leaves nozzles,

degrees; F_{min}, M_{max} are the minimum and maximum nozzle cross-sections; η_u, η_e are the peripheral and actual efficiency of the stage; l_c is the nozzle height, mm; L_1, L_2, L_3 are the moving blade heights of the 1st, 2nd, and 3rd rows, mm; L'_H, L''_H are the stator blade heights of the 1st and 2nd rows, mm; q is the degree of nozzle divergence.

Modern auxiliary turbines, as seen in table 1, are distinguished by the low power values (15-300 kWt, rarely to 700 kWt), characteristics u/c_1 = (0.1-0.2), admission (0.025-0.25); high supersonic steam velocities (M_c1 = 1.8-3.5); moderate speed values (to 12,000 per minute), blade rim speed (100-200 m/sec), actual efficiency (0.26-0.40, and in isolated cases to 0.46). Presently, the main stage type is the pressure stage with two speed stages (double-rim wheel), but a triple-wheel (pressure stage with three speed stages) is also used.

The use of triple-rim wheels is explained by the fact that their peripheral efficiency in the area u/c_1 < 0.13 is 10-15% greater than double-rim wheels. But table 1 shows that modern two-rim wheels provide the same or greater stage efficiency as three-rim wheels. This may be attributable to the fact that in the case of three-rim wheels with high peripheral efficiency the losses from incomplete admission, leaks, etc. are greater than in the case of two-rim wheels (see table 2, columns 2 and 3, values η_u and η_e).

Returning to table 1, we see that values u/c_1 < 0.10 are not encountered. In reference /2/ the field of use of three-rim wheels with initial steam parameters of 23 kg/cm^2 and 360°C is determined thusly: when N_e = 20 hp field u/c_1 < 0.14; when N_e = 120 hp field u/c_1 < 0.145; when N_e = 400 hp field u/c_1 < 0.15. This means that three-rim wheels may acutally be used in the range u/c_1 ≈ 0.10-0.13, because the boundary field can efficinetly be given to a two-stage wheel as lighter and cheaper.

On the basis of what has been presented further use of three-rim wheels must be completely suspended, and efforts concentrated on the improvement and unification of double-rim wheels as the main element of modern auxiliary turbines.

The situation with economy and comparison of stages with one and two speed stages appears somewhat different. In the case of modern two-rim wheels of auxiliary turbines the average rim speed of the blades is within the limits of 100-200 m/sec, while the parameter u/c_1 is in the limits of

Table 1
Main Parameters of Modern Marine Steam Auxiliary Turbines

Symbol	Type of turbine stage		
	3-rim speed stage	2-rim speed stage	single rim with rotating chamber
N_e	50—125	50—750	15—52
n	2000—3600	3600—10 500	7000—12 000
P_1	20—25	20—58	23—25
t_1	250—340	280—470	250—340
P_2	2—2,8	0,5—3	1,5—2,5
α_1	16—18	15—20	15—20
P_1/P_2	7—12,5	10—32	10—12,5
l_c	9—10	10—12	8—20
$H_{a\kappa}$	80—120	100—200	100—120
u_1	70—100	100—200	75—190
u/c_1	0,10—0,126	0,10—0,20	0,17—0,21
ε	0,06—0,160	0,025—0,250	0,05—0,157
D	450—540	250—650	200—300
G	1500—4200	1000—7000	380—1260
F_{max}/F_{min}	1,6—3,2	1,9—4,5	2,1—2,5
M_{c_1}	1,8—2,4	1,8—3,5	1,8—2,4
η_e	0,275—0,340	0,26—0,46	0,37—0,45
No. of machines in table	10	25	5

0.10-0.20. Given such u/c_1 values the peripheral (rim) efficiency of a two-rim wheel can exceed that of a single-rim wheel by 20-45%. Losses from incomplete admission (ventilation, edge, etc.) is less in single-rim wheels, but it is sufficient to compensate the indicated difference. Given modern heads, rim speeds, and other characteristics, the pressure stage with two speed stages in the form of a two-rim wheel retains its importance. The importance of improving this kind of stage is evident. Table 3 gives the main parameters of individual Soviet designs of two-rim marine auxiliary turbines. Power losses in these turbines are 54-74% of the adiabatic drop; their distribution in the design mode is lower applicable to the turbine data on which is given in table 3, column 4 (calculated data).

Table 2

Main Parameters of Four Turbine Stages (Design Mode)

Symbol	Type of turbine stage: 3-rim speed stage	2-rim speed stage	2-rim speed stage	1-rim stage rotating chamber
N_e	65	62	68	60
n	3600	3500	7850	11 800
P_1	23	25	38	21
t_1	340	$x = 0{,}95$	290	$x = 0{,}995$
P_2	2	2	3	2
ε	0,060	0,088	0,064	0,120
α_1	18°30′	15°	14°50′	16°
l_c	10,6	10,0	10,0	10
L_1	14	12	16,2	14
$L'_н$	18	14	21,2	—
L_2	23	16,5	25,0	—
$L''_н$	28	—	—	—
L_3	33	—	—	—
D	540	540	400	300
P_1/P_2	11,5	12,5	12,6	10,5
$H_{ад}$	102,8	99,9	115,9	97,3
G	1650	1600	1570	1178
u/c_1	0,11	0,109	0,168	0,216
η_u	0,416	0,380	0,475	0,654
η_e	0,329	0,336	0,293	0,448
Stator blades				
1-й ряд	СЛ153327	20ТР2Б	СЛ153030	—
2-й »	СЛ154840	—	—	—
Moving blades				
1-й ряд	СЛ152525	20ТР1В	ФЛ152622	Старый профиль
2-й »	СЛ154038	20Т3	ФЛ154840	
3-й »	СЛ154949	—	—	—

Table 3

Main Parameters of Separate Designs of Two-Rim Marine Pump-Drive Auxiliary Turbines

Symbol	Feed pump	Feed pump	Feed pump	circulation pump
N_e	49	406,7	134	230
n	7600	4300	5400	10 27[illegible]
P_0	40	35	40	25
t_1	290	435	290	$x = 0,95$
P_2	3	2,5	3	0,[illegible]
α_1	14°50′	20°	15°	18[illegible]
l_c	10	10	10	10[illegible]
$H_{ад}$	115,9	149,2	102,1	138
u	159	146	150	215
ε	0,0635	0,250	0,118	0,55[illegible]
D	400	650	530	400
G	1275	6050	2500	3800
η_u	0,485	0,427	0,613	0,64[illegible]
η_e	0,293	0,379	0,449	0,45[illegible]
u/c_0	0,169	0,131	0,163	0.20
P_1/P_2	10	13,2	9,4	28,5
M_{c_1}	1,62	1,64	1,61	3,45
1st row of moving blades	ФЛ152622	СЛ152525	20ТР1В	ТР1Б
stator blades	ФЛ153030	СЛ153327	20ТР2Б	ТР3Б
2nd row of moving blades	ФЛ154840	СЛ154033	20ТРЧА	ТРЧА
L_1	16,2	16,5	11,5	15
$L'_н$	21,2	23,5	15	19
L_2	25	29,5	19,15	27
Year of construction	1958	1955	1960	1959

Type of Loss	Loss in % of H_{ad}
In nozzles	9.8
In moving blades of 1st row	18.0
In stator blades	5.0
In moving blades of 2nd row	1.57
From moisture	3.90
Friction and ventilation	6.50
Edge	2.35
End	4.53
With outlet velocity	2.35
Mechanical	1.10
Total	55.1

The following main possibilities exist for the improvement of a two-speed stage of this kind:*

* Such questions as the selection of steam parameters and the counter-pressure of the auxiliary turbines, the combining of several auxiliary engines into one unit for driving from one turbine, etc. are not examined here. These questions pertain to the theory and design of steam turbine plants as a whole.

1) developing profiles of supersonic moving blades of the first rim at large M_{c_1} = 1.8-3.5 numbers for condensing turbines M_{c_1} up to 4.5 (inasmuch as the losses in the first rim of the moving blades are ≈20%, even in cases of new supersonic profiles) /3/;

2) correct selection of designs of supersonic nozzle systems and their geometric parameters (α_1, q, thickness and form of edges, profile of nozzles, etc);

3) improvement of design of blading section for purpose of reducing losses from incomplete admission and the like (compaction on admission arc, compaction of nozzle segment, accounting for specific design features of admission stage, etc.) /4/;

4) development of new computational method and correct selection of optimum parameters of supersonic two-rim admission stage.

For a two-speed wheel it is desireable in several cases to bring u/c_1 to 0.20-0.22, i.e., to increase u/c_1 by 20-100% against existing values.

This can be achieved by a sharp increase in the rim speed, shift to other designs of two-speed wheels, or through the use of two or three pressure stages.

Increasing the rim speed by increasing the diameter is not desireable. A serious obstacle to increasing rim speed is the heavy wheel rim. But a two-speed stage can also be designed on the basis of a single-rim with re-supply (SPP or Kienast stage), Fig. 1.

Here a light single-rim wheel permits the rim speed to be increased by a factor of 1½-2, using the same materials, i.e., increase u/c_1 and bring it to its optimum values. Re-supply of the working substance permits the admission degree to be increased and, consequently, lessen losses connected with partial admission. With increased rim speed losses due to disc, band, and ventilation friction increase, although the relative growth of these losses will be less than the gain from reducing losses with the outlet speed and from ventilation loss (associated with increasing admission). The value of the additional losses characteristic of stages with re-supply (overflows, rotating chamber) has not yet been explained.

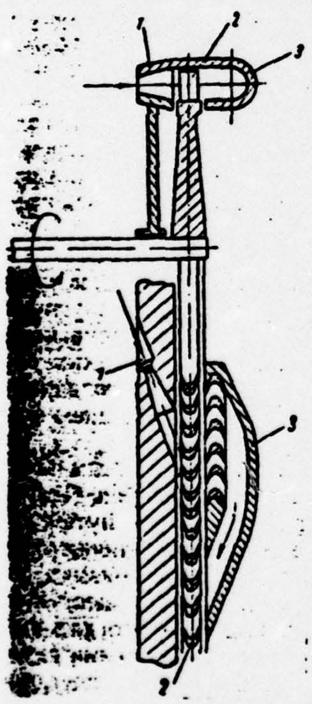

Fig. 1. Single-rim two-speed stage with re-supply.

1 - nozzle; 2 - moving blades; 3 - rotating chamber

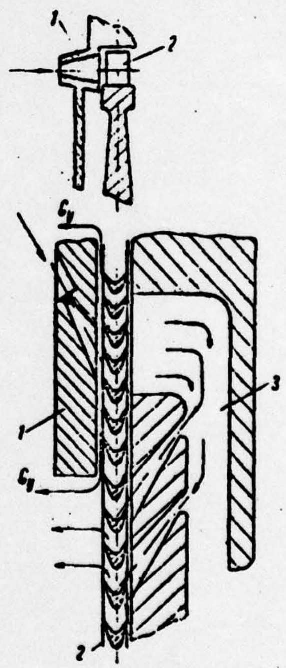

Fig. 2. Pressure stage with re-supply (steam supply from two sides). 1 - nozzle; 2 - moving blades; 3 - rotating chamber.

The speed stage with re-supply is simple in design and is advantageously used in cases of small admission degree values when the ventilation loss is significant. Analysis shows that in this stage the nozzles can occupy up to 0.3 of the entire circumference. In this case rather great powers necessary for driving all auxiliary machinery encountered can be used.

Separate comparisons of the two-rim wheel and the single-rim two-speed stage with re-supply (see tables 1 and 2) show that in the case of equal powers and other comparable conditions the efficiency of the single-rim two-speed stage with re-supply is considerably (30-40%) greater than that of a stage with a two-rim wheel. It must be noted that in planning the turbine stages of auxiliary machinery the single-rim two-speed stage with re-supply is very rarely used. Due to the lack of theoretical and experimental data there is no rational method of planning this stage type. Presently, to improve a stage with re-supply the greatest effect is produced through aerodynamic investigation and improvement of rotational chambers, investigation of steam leaks, selection of the optimum admission degree and blade height.

The use of two- and three-stage turbines instead of single-stage would permit a substantial increase in u/c_1 (by 40% for two pressure stages,

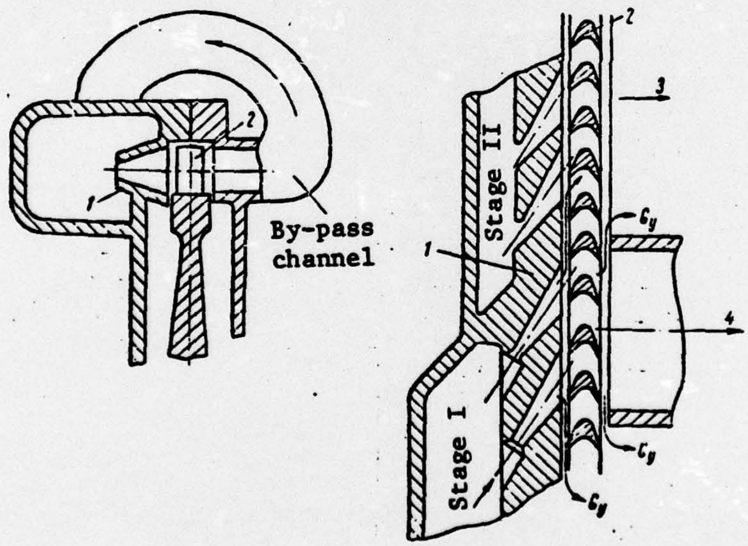

Fig. 3. Pressure Stage with Re-Supply (Steam Supply from One Side).

1 - nozzle; 2 - moving blades; 3 - outlet from stage II; 4 - entrance to stage II through by-pass channel.

70% for three, about double for four stages). However, due to weight and size limitations it is impossible to switch over to 2-3 stage disc-diaphragm design in most cases. But it could be possible on the basis of a pressure stage with re-supply, as shown in Figs. 2 and 3.

Among the advantages of the given designs are:

1) The possibility of obtaining optimum u/c_1 and, consequently, high values of rim efficiency of the stage in connection with a reduction in the heat drop on one pressure stage and with the possibility of a significant increase in rim speed of the light, thin disc;

2) Lesser loss for ventilation in connection with admission stage; countering this are the additional losses from leaks in rim gap and from flow turning 180° in the rotating channels;

3) Small size, simplicity of design, and low manufacturing costs since the number of blades is reduced by a factor of 2-3 and the rotor weight is greatly reduced.

The question of the desireability of going over to single-stage turbines with one speed stage (active single-rim stage) is of considerable practical interest.

In reference /2/ for initial steam parameters of 23 kg/cm^2 and 360°C and u = 120-180 m/sec, the use of such turbines is considered desireable if u/c_1 is greater or equal to the values given in table 4.

Table 4

Minimum Values of u/c_1 for Single-Rim Active Stages

N_e	D		
	300	500	700
15	0,210	0,225	0,240
75	0,250	0,265	0,270
300	0,275	0,275	0,275

As we see, in order to provide general use of the active single-rim stage and end the manufacture of turbines with speed stages, it is enough to raise the rim speed by a factor of 1.5-2. The use of active single-rim stages when there are small heat drops can be desireable even with modern rim speeds.

When values u/c_1 > 0.28 the outlet loss in the single-rim stage will

be less than in the two-rim. Losses in the cascades are reduced ~6-8% from the heat drop, since only one cascade instead of three remains. The losses from incomplete admission become less for the same reason, but increase somewhat due to the increase in u/c_1. The increase in rim speed and rpm's results in some increase in disc and band friction. On the whole, the balance will favor the single-rim stage, but this question demands special study.

The introduction of active single-rim stages can also be effected by other methods: leave the rim speeds at the present level and build active single-rim stages with diffusers. The outlet loss is 2-3% of the adiabatic heat drop for the two-rim stage and would be 15-50% for the single-rim. Under these conditions the use of a diffuser would permit the efficiency of the single-rim stage to be increased substantially, especially in the case of low u/c_1 values. To a lesser degree this pertains to two-rim wheels.

A two-rotor stage would permit a high stage efficiency to be obtained with today's moderate rim speeds. However, this stage is considerably more complicated than the other types of stages examined above, and its use at sea is made more difficult by the need to match the characteristics of the two different machines, not just in the rated but in other modes as well. Because of this the two-rotor stage may be of interest in the relatively rare cases of driving two one-mode machines with similar power and rpm relationships for the given stage.

As has been noted above, the progressive approach in the development of auxiliary turbines would be to increase blade rim speed by increasing turbine rpm's. In the presence of modern, comparatively quiet-running machines (pumps, generators, etc.) this means increasing the gear ratio. For presently used gears with parallel axes this is the same as increasing the gear size and weight. Therefore making the transition to faster-operating actuators remains an urgent task.

With any machinery, the important way of reducing the size and weight of the machine is to go over to the use of planetary gears. In the case of modern, low power, planetary gears, gear ratios from 1 to 60 and gear efficiencies of 0.04-0.99 are possible. The switchover from conventional to planetary gears can provide a weight reduction by a factor of 2-6 /5/. Greater values are related to those cases where the transition to planetary gears is possible to increase the gearing load. This was not possible or feasible for conventional gears or two-stage gears. Coaxiliality of turbine, reduction gear, and machinery makes it possible to dispense with the traditional auxiliary machinery; the turbo-reduction gear approaches the modern auxiliary gas turbine engine with respect to compactness and weight.

Increasing the rim speed of two-rim and one-rim wheels to 250-400 m/sec can be completely provided for by existing, already proven disc materials /6/. In modern turbo construction heavily loaded discs, long, heavy lifting

blades with a rotational rim speed to 400 m/sec are already quite widely used. Calculations show that the stresses in discs of auxiliary turbines will always be considerably less.

Selection of the characteristics of auxiliary turbines at the present time does not ensure correct matching of thermodynamic and geometric parameters in a single optimum set that would ensure maximum efficiency. For example, selection of the optimum u/c_1 is made on the basis of the comparison of many variants that differ in diameter or rpm's. That variant in which the sum of rim losses and friction and ventilation losses, and edge loss is minimum is considered optimum. In this, disruption of the geometric similitude of the variants (relationship between nozzles and wheel diameter, length of admission arc, etc.) are ignored, while the selection of the loss coefficients is in no way connected to the geometry of the stage, to M, Re and other criteria.

In this approach the concept of optimum variant is illusory and stage efficiency is inevitably lower than would be the case given an optimum set of parameters. The existing method of calculation is still not able to determine this optimum set of parameters.

In this regard attempts to introduce into the calculations the so-called power-speed coefficient $n_s = n \frac{\sqrt{G}}{H_{ad}^{0.75}}$ and equivalent diameter $D_s = D \frac{H_{ad}^{0.25}}{\sqrt{G}}$ /7, 8/.

It may be shown that for given stage geometry and M and Re criteria, i.e., for given losses in the nozzles and blades fixed by the coefficients φ and ψ /9/, the values n_2 and D_s are unambiguously determined by stage characteristic u/c_1. But in addition, the number of revolutions n enters directly in n_s, while the most characteristic geometric dimension D is included in D_s, i.e., the values used in the process of designing turbines.

The new element is that the stage efficiency and geometric relationships (for example, relationship of nozzle height or size of blade channel throat to wheel diameter) are expressed in terms of n_s and D_s. The construction of efficiency curves in the n_s-D_s axes and the superimposition on this field of the curves representing the geometric relationships make it possible to select optimum (providing greatest efficiency) n_s and D_s values immediately, and to note those geometric relationships and u/c_1 that must be observed in order to obtain the desired maximum stage (turbine) efficiency. Knowing n_s and D_s, it is easy to determine the number of turbine revolutions n and the wheel diameter for known heat drop and steam flow.

Thus, n_s and D_s become the connecting link between the stage geometry and its efficiency and the basis for determining the dimensions and number of turbine revolutions. There is the possibility for a matched selection of the optimum set of turbine parameters.

The n_s-D_s diagram noted is characterized by its clarity of representation. Significantly more important is that this method of selecting parameters results in the auxiliary turbine efficiency being from 15 to 100% and more higher /8/. These figures are possibly exaggerated, but the new method of selecting parameters is promising.

The construction of n_s-D_s diagrams and their use in designing auxiliary turbines requires that the nature of the relationship between geometric parameters and stage losses be made more precise. This again indicates the importance of studying these questions applicable to a supersonic stage with low admission and segmental nozzles.

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